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STRUCTURAL AND THERMAL ANALYSIS OF DISC BRAKE USING SOLIDWORKS AND ANSYS

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ABSTRACT

Braking system represents one of the most fundamental safety critical components in modern vehicles. Brake absorbs kinetic energy of the rotating parts (Wheels) and the energy is dissipated in the form of heat energy to the surrounding atmosphere. It decelerates or stops the vehicle. When brake is applied to the disc brake it is subjected to high stress, thus it may suffer structural and wear issues. Hence for the better performance, structural, stress and the thermal analysis is preferred to choose low stress material. The objective of this paper is to model and analyse stress concentration, structural deformation and thermal gradient of disc brake. Here the disc brake is designed by using Solidworks and analysis is done by ANSYS workbench R 14.5.

Key words: Solidworks, ANSYS, Disc Brake, FEA (Finite Element Analysis)

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1. INTRODUCTION

A brake is a device which uses frictional resistance to stop motion of machine or vehicle. The brakes absorb kinetic energy and dissipate it as heat energy. Brake systems must have following requirements:

- Vehicle must stop within a minimum distance in emergency.
- Braking properties must not fade with constant prolonged application.
- It must have anti wear properties.[1]

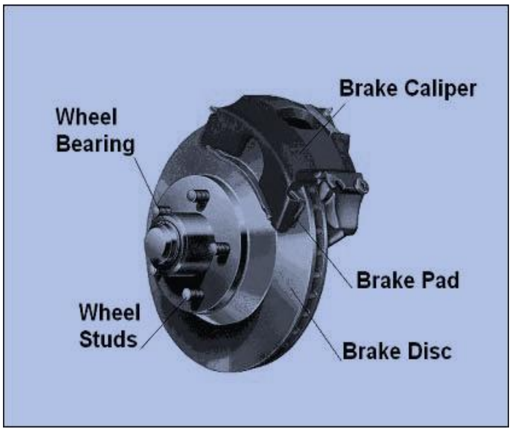


Figure 1 Source: Google.com, Shows Disc Brake Assembly [19]

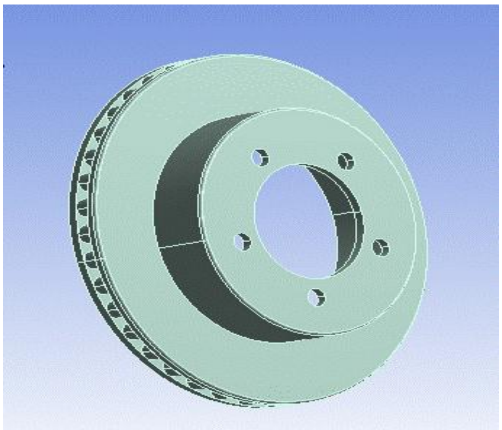


Figure 2 Shows CAD Model of Disc brake

Disc brake is an important component of vehicle retardation system. It is the type of brake that uses callipers to squeeze pairs of pads against a disc to create friction. That friction slows the rotation of a shaft (vehicle axle) to hold it stationary or slow its rotational speed. Disc brake is usually made of cast iron or ceramic composite such as carbon, aluminium, and silica. Friction material known as brake pad is made to engage on both side of disc either mechanically or pneumatically. This friction material causes disc to slowdown or stop. [2]

The disc brake is sandwiched between two pads activated by a cylinder backed in a calliper mounted on the stud shaft. At the point where the brake lever is pressed, pressurised hydraulic pressed fluid is constrained in the chamber pushing the contradicting cylinders and brake parts. [12] When the pad is pressed against the disc, the brake absorbs kinetic energy of the vehicle and it is transferred into heat which is mainly absorbed by rotor and brake pad. This heat is dissipated into the surrounding atmosphere. [3] Due to the generation of frictional heat on the interface of the disc and pad, there is rise in temperature. When this temperature exceeds the critical value of the given material, it leads to catastrophic events such as brake fail, premature wear, failure of bearing, thermal crack or vaporisation of brake fluid.[4] Furthermore, due to heat generation at the disc pad interface, global deformation occurs in disc and pad. Some common deformations are coning and buckling [13-15]

Besides this, macro cracks may appear on the disc brake in radial direction after some brake cycles. Thus life and performance of the disc brake is affected. [16-17]

Lee and Yee [4] found out that the uneven distribution of temperature on the surface of frictional pad causes thermal distortion popularly known as coning and this is responsible for judder effect and disc thickness variation.

Mainly three types of mechanical stress is subjected on disc brake.

- Traction force, caused by centrifugal effect and it occurs when wheel is rotating and no brake force is applied to the disc.
- Compressive force, when the brake is applied due to action of the force, exerted by pressing the pad perpendicular onto the surface of the disc.
- Due to braking action caused by rubbing on the brake pad against the surface of the disc. It acts in opposite direction of the disc rotation.[5]

Dufreny and Weichert [18] presented the existence of radial tensile stress on disc surface by measuring with hole drilling strain gauge method. The effect of the pressure distribution plays a vital role. Uniform pressure distribution between pad and the rotor leads to uniform pad wear and even friction coefficient. On the other hand, non-uniform pressure distribution may lead to uneven wear and its called disc brake squeal. [6] Frictional heat generated between two sliding bodies is responsible for thermo-elastic deformation which ultimately alters the contact pressure distribution. In order to predict the temperature distribution, many investigation had been carried out on heat generation phenomenon between contact surfaces in brake. Due to the characteristic pressure sliding speed, coefficient of viscosity this process is very complex. Finite element analysis is used to solve it. [7].

From experimental setup it has been found out that coefficient of viscosity generally decreases with increasing sliding speed and applied load but increases with increasing disc temperature of 230 degree and then decreases with above disc temperature. Furthermore specific wear rate was found to increase with increase in sliding speed and disc temperature. [8]

In addition if sliding speed is high, it results in instability in thermo mechanical. Besides this, it leads to non uniform contact distribution, which produces hotspot. This is accompanied by high local stress that can lead to material degradation. [9] Due to repetitive braking, temperature of ventilated disc can rise relatively faster than solid disc [10]. Ventilated disc is lighter, it has convective heat transfer and can control the temperature rise, and the effect of thermal problem. However ventilated disc due to uneven temperature around the disc, there may be increase in judder problem. [11]

Valvano and Lee carried out a study on the technique to determine thermal distortion of brake rotor. Due to the severe thermal distortion of brake, it affects system response and brake judder propensity. Accurate prediction of thermal distortion improves the disc brake design [12]

In the braking phase temperatures and thermal gradient are very high. It causes stresses and deformation which is indicated by its appearance, cracks on the disc [13]

2. OBJECTIVE

The objective is to design a Disc brake using Solidworks 12.0 and carry out the finite element analysis (FEA) on the prepared model using ANSYS 14.5. Thus we obtained the values of shear stress, total deformation, convective heat transfer coefficient and temperature distribution on disc brake.

3. DESIGN AND CALCULATION

Automobile Model: PORSCHE CAYMAN (2.7)

Table Geometrical dimensions and application parameters of braking

Item	Values
Disc Diameter	298 mm
Disc Thickness	24 mm
Overall Height of automobile	68 mm
Centre Diameter	98 mm
Size of pad	114*78*16
Weight of Automobile	1330 kg
Top Speed of Automobile	160 mph
Tire Size	235/45ZR18
Effective Radius of Rotor, R_r	110 mm
Mass of the disc	5 kg
Specific heat, C_p	450 $Jkg^{-1}k^{-1}$
Deceleration	12.9 ms^{-2}

Nomenclature

F_d = Force on the disc

R_t = Radius of tire

R_r = Radius of rotor

R_2 = Outer Radius of the pad

R_1 = Inner Radius of the pad

t_s = time taken to stop the automobile

v = initial speed

v = $\sqrt{\mu g d}$ = 15.002 m/s

Stopping distance: $\frac{V^2}{2a} = \frac{(160x1000/3600)^2}{2x12.9} = 76.56m$

Rotational Speed (ω) = $\frac{v}{R_t}$ = 44.91 rad/sec

t_s = 3.44 s

Kinetic energy (K.E) = $\frac{1}{2}mv^2$ = 149664.9027 Joule

Heat generated (Q) = $mC_p\Delta t$= 7740 Joule

Restoration Energy (R.E) = 3% of kinetic energy = 4489 Joule

Total energy = K.E + R.E = 154154.8498 Joule

Area of rubbing face = total energy × (R_2- R_1) = 15415484.98 mm^2

Force on disc (F_d) =$\frac{(30\%)*\frac{1}{2}mv^2}{2*\frac{R_r}{R_t}(v\circ t_s-\frac{1}{2}\left(\frac{v}{t_s}\right)*t^2s)}$=2641.7669N

External pressure between disc and pad

= force applied to the disc

= $\frac{F_d}{Ac*\,\mu} = \frac{2641.7237}{8892*0.3}$ = 0.99≈ 1 MPa

Heat flux = heat generated on each front wheel = heat generated /time to stop vehicle = 36273.633 w/m^2

Disc usable area= $\frac{\pi}{4}({R_2}^2 - {R_1}^2)$ = 0.0622 m^2

4. MATERIAL PROPERTY

For analysis we have consider **Aluminium Alloy 6262 T-9**

Density	2.72 g/cm^2
Young Modulus	70000 MPa
Poisson Ratio	0.33
Ultimate tensile Stress	400 MPa
Isothermal thermal conductivity	171 w/$m^{-1}k^{-1}$
Specific Heat	890 J/kgk^{-1}

5. METHODOLOGY

5.1. Modelling in ANSYS CFX

Air flow characteristics vary significantly with under body structure, component shape. We have used average heat transfer coefficient which is calculated from the measured cooling coefficient by an iteration algorithm in order to carry out thermal analysis.

Here we took one quarter of disc, and then we defined the field of air surrounding the disc. After obtaining the model on CFX PRE and specifying the boundary conditions, then we begin the calculation on CFX. Figure 3 shows the elaborate CFD model which will be used in ANSYS CFX PRE.

After declaring the physical characteristics of fluid and solid, we take the following temporary condition in order to determine the temperature field in a disc brake during braking phase

- Braking time= 3.5 (S)
- Incremental time= 0.01 (S)
- Initial time = 0 (S)

ANSYS CFX solver automatically calculates heat transfer coefficient at the wall boundary which is shown in Figure 4.

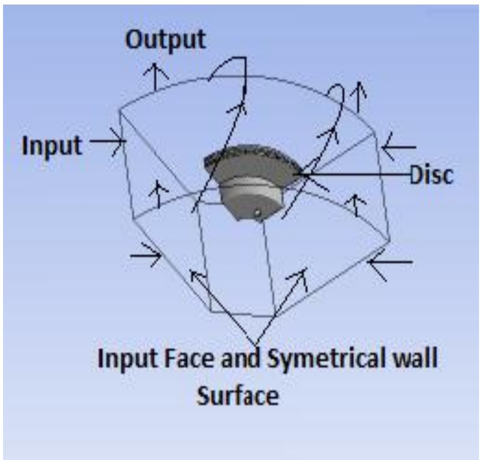


Figure 3 Shows CFD Model of Disc Brake

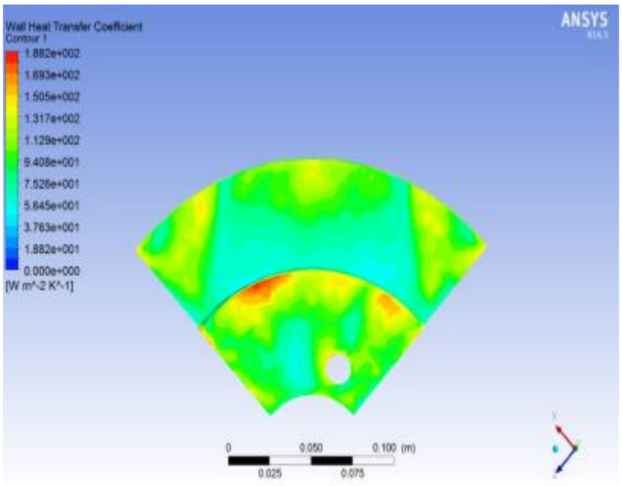
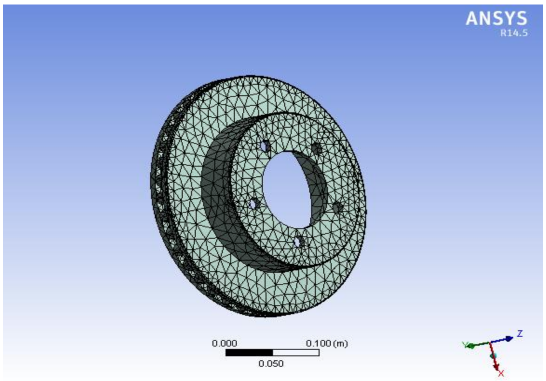


Figure 4 shows Heat Transfer coefficient at the wall of Disc Brake

5.2. Meshing of the disc

Figure 5 shows the meshed model of disc brake for Thermal analysis process. For analysis Ventilated disc brake was messed using triangular surface mesher. The number of Nodes used in this meshing is 95189 and elements are 65016. The model is mashed and analysed to get the detail result of contact zone (disc-pad). This is vital because in this zone the temperature rises significantly.

Figure 6 shows the meshed model of disc brake for Structural analysis process. For analysis Ventilated disc brake was messed using triangular surface mesher. The number of Nodes used in this meshing is 43161 and elements are 28042. So, the model is meshed and then analysed to get the detail and correct result of the stress of the contact surface



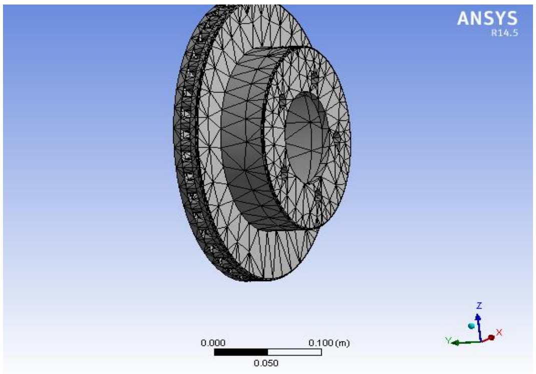


Figure 5 Shows Thermal meshing of Disc Brake **Figure 6** shows Structural meshing of Disc Brake

5.3. Loading and Boundary Condition

For thermal analysis, the temperature distribution depends upon the heat flux entering the disc through both sides of the disc and wall heat transfer coefficient. For analysis the initial and boundary conditions are introduced in the transient thermal nodule of ANSYS WORKBENCH. The conditions for numerical analysis are as follows:

Number Of Steps	180.
Current Step Number	1.
Step End Time	1. s
Auto Time Stepping	Off
Define By	Time
Time Step	0.25 s
Time Integration	On
Initial temperature of disc	65 ℃
Material of disk	Aluminum 6262-T9

For structural analysis, temperature and corresponding stress in disc brake vary under freeway driving conditions. For analysis the initial and boundary condition are introduced in structural module of ANSYS workbench. Initial and boundary conditions area as follow

Initial temperature of disc	65 ℃
Pressure applied on both surface	1 MPa
Rotational speed of disc	44.91 rad/s

5.4. Finite Element Method (FEM)

It is a numerical technique for finding the approximate solutions to boundary value problems for partial differential equations. It uses subdivision of a whole problem domain into simpler part, called finite elements and solve the problem by minimizing an associated error function. The subdivision of the whole domain has several advantages:

I. Accurate representation of the complex geometry.
II. Inclusion of dissimilar material property.
III. Easy representation of the solution.
IV. Capture of the local effects.

It divides the domain into a group of sub domain; every sub domain is represented by a set of element equations of the original domain.

6. RESULTS AND ANALYSIS

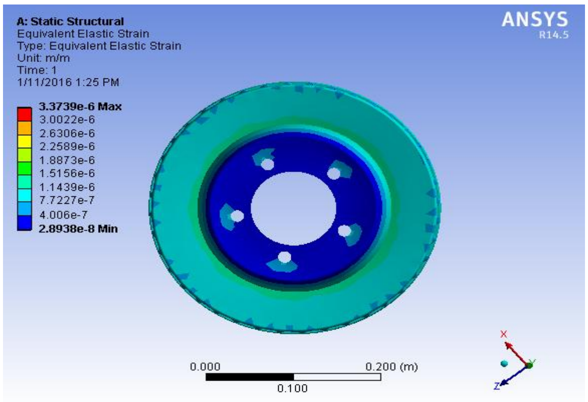


Figure 7 Shows Equivalent Elastic Strain of Disc Brake

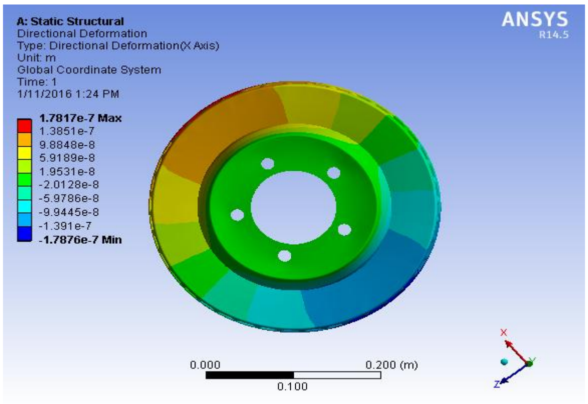


Figure 8 Shows Directional Deformation of Disc Brake

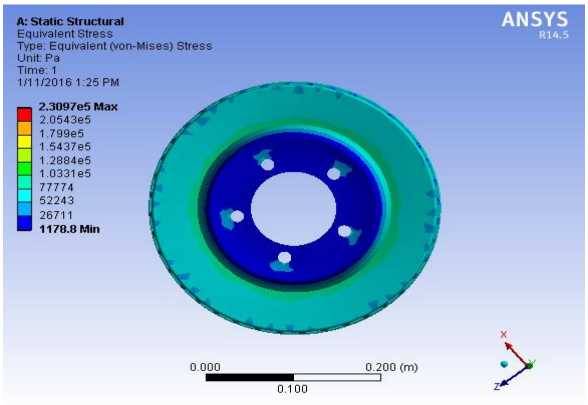


Figure 9 Shows Equivalent (Von-Mises) Stress of Disc brake

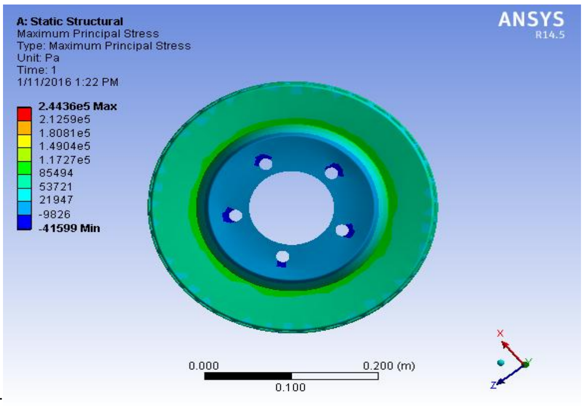


Figure 10 Shows Maximum Principal Stress of Disc brake

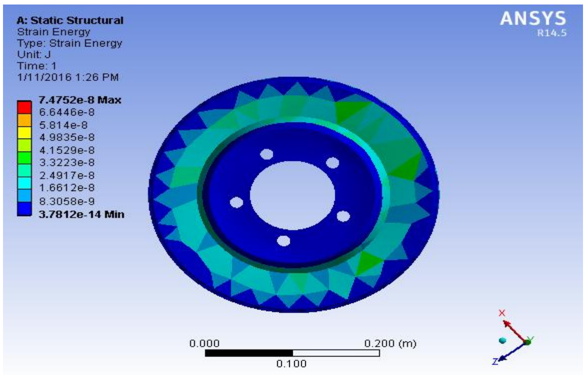


Figure 11 Shows Strain Energy of Disc Brake

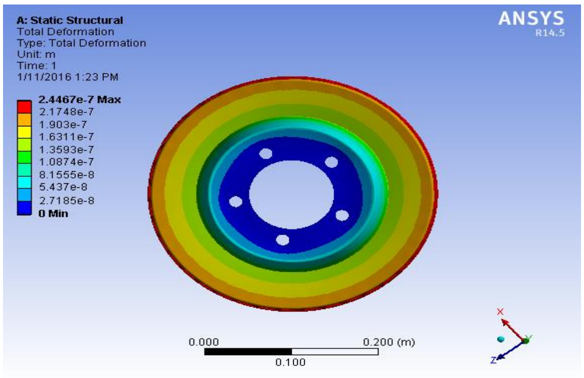


Figure 12 Shows Total Deformation of Disc Brake

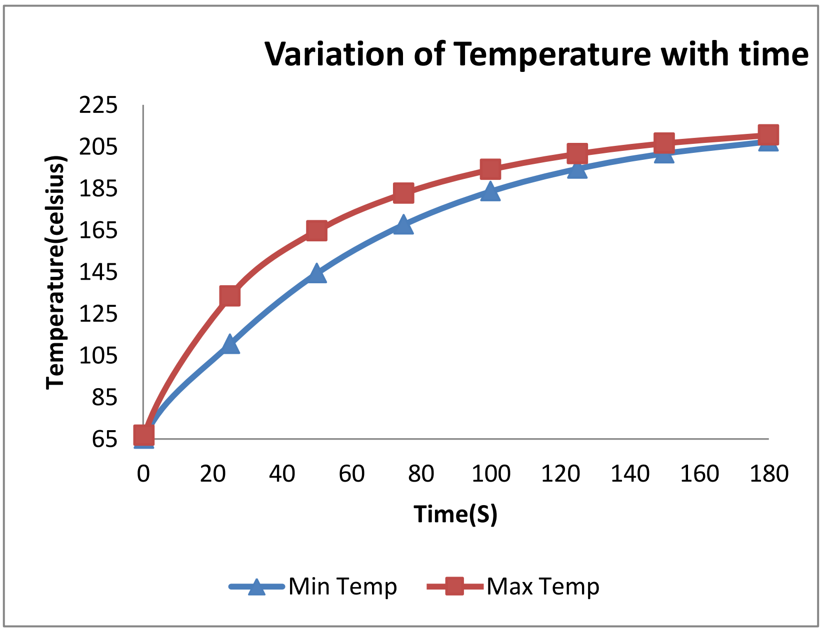


Figure 13 Shows Variation of Temperature with time of Disc Brake

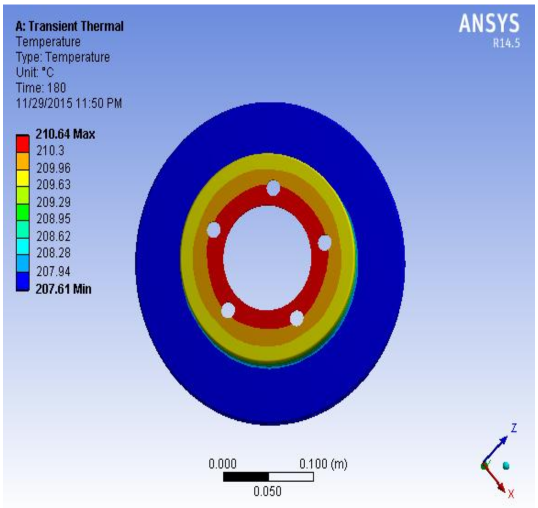


Figure 14 Shows Temperature distributions on Disc Brake

RESULT

S. No	Parameter	Maximum Value	Minimum Value
(a)	Equivalent Elastic Strain	3.379e-6 (m/m)	2.8938e-8 (m/m)
(b)	Directional Deformation	1.7817e-7 (m)	-1.7876e-7 (m)
(c)	Equivalent (Von Mises) Stress	2.3097e5 (pa)	1178.8 (pa)
(d)	Maximum Principal Stress	3.4436e5 (pa)	-41599 (pa)
(e)	Strain Energy	7.4752e-8 (J)	3.7812e-14 (J)
(f)	Total Deformation	2.4467e-7 (m)	0 (m)
(g)	Temperature	210.64 (‘C)	207.61 (‘C)

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